ADA

LABORATORY BASED AXLE LUBRICANT EFFICIENCY EVALUATION

INTERIM REPORT
TFLRF No. 459

by
Robert W. Warden
Edwin A. Frame
Adam C. Brandt
Scott J. Tedesco

U.S. Army TARDEC Fuels and Lubricants Research Facility
Southwest Research Institute® (SwRI®)
San Antonio, TX

for
Allen S. Comfort
U.S. Army TARDEC
Force Projection Technologies
Warren, Michigan

Contract No. W56HZV-09-C-0100 (WD02)

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EXECUTIVE SUMMARY

Laboratory based evaluations of HMMWV axles showed the ability to discriminate between oils on efficiency, providing the groundwork for developing an axle efficiency test stand suitable for military purposes.

FOREWORD/ACKNOWLEDGMENTS

The U.S. Army TARDEC Fuel and Lubricants Research Facility (TFLRF) located at Southwest Research Institute (SwRI), San Antonio, Texas, performed this work during the period January 2011 through July 2014 under Contract No. W56HZV-09-C-0100. The U.S. Army Tank Automotive RD&E Center, Force Projection Technologies, Warren, Michigan administered the project. Mr. Eric Sattler (AMSRD-TAR-D/MS110) served as the TARDEC contracting officer's technical representative. Mr. Allen S. Comfort of TARDEC served as project technical monitor.

The authors would like to acknowledge the contribution of the TFLRF technical and administrative support staff.

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ACRONYMS AND ABBREVIATIONS

%	Percent
˚C	Degrees centigrade
˚F	Degrees Fahrenheit
S	Seconds
ASTM	American Society for Testing and Materials
CAT	Caterpillar
cSt	CentiStoke
FMTV	Family of Medium Tactical Vehicles
GO	Gear Oil
GVW	Gross Vehicle Weight
HDO	Heavy Duty Oil
lbs	Pounds
mph	Miles Per Hour
Nm	Newton-meter
OE	Oil Engine
OEA	Oil Engine Arctic
SAE	Society of Automotive Engineers
TM	Technical Manual
ARL	Army Research Lab
HMMWV	High-Mobility Multipurpose Wheeled Vehicle
PLS	Palletized Load System
HET	Heavy Equipment Transporter
PEVEL	Power and Energy Vehicle Environmental Laboratory
MRAP	Mine Resistant Ambush Protected

1.0 BACKGROUND AND OBJECTIVE

The U.S. Army desires to increase the fuel efficiency of its ground vehicle fleet. One potential area for fuel consumption improvement is the lubricating fluids located throughout the driveline. By improving the lubricating fluids used, a reduction in mechanical losses can be achieved [1]. These mechanical losses can occur in many forms including frictional, pumping, and churning losses, and are dependent on the fluid's chemical/physical properties and equipment design. A relatively small increase in driveline efficiency could have a significant impact financially when multiplied over the entire U.S. Army vehicle fleet. A previously reported investigation looked at the fuel consumption effects of engine, transmission, and axle gear lubricants as used in the Family of Medium Tactical Vehicles (FMTV) [3]. Fuel consumption changes were determined based on the SAE J1321 Fuel Consumption In-Service Test Procedure – Type II [4].

Figure 1: Test FMTVs

This report covers the second phase of the work directive that investigated the feasibility of a laboratory based method for determining efficiency gains from axle lubricants, and provides a preliminary axle test stand design that could be used for Army truck axles of varying sizes.

2.0 LABORATORY BASED AXLE OIL EVALUATION

2.1 TEST CYCLE AND STAND

The TARDEC Fuels and Lubricants Technology Team was informed by the Army Research Lab (ARL) of a program to evaluate High-Mobility Multipurpose Wheeled Vehicle (HMMWV) differentials for efficiency impacts that was being conducted at Southwest Research Institute. The ARL program was focused on determining the effects of axle component coatings on axle efficiency. The TARDEC project examined the effects of different lubricants on axle efficiency. This provided an opportunity to share stand development efforts for multiple U.S. Army programs. The stand configuration was a "T-style" layout, consisting of one motoring input to the differential, and two power absorption units on each output independently. An enclosure surrounded the axle differential to allow for temperature control when required. This is shown in Figure 2 and Figure 3.

Figure 2: Axle Differential T-Style Stand

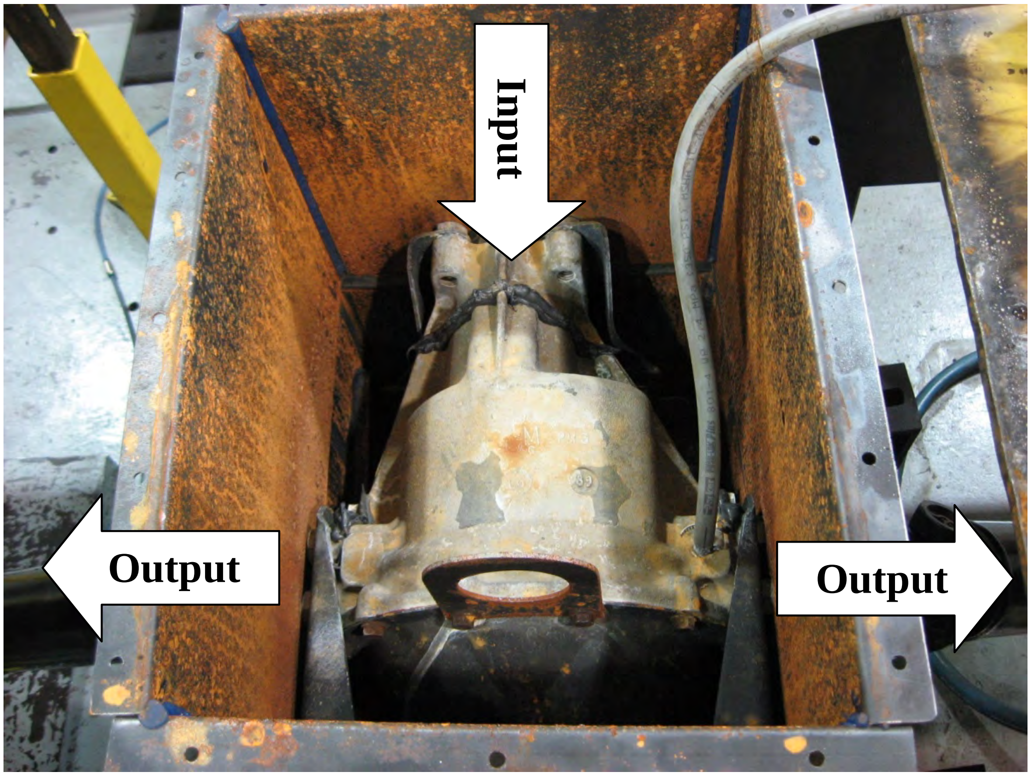

Figure 3: HMMWV Axle Differential in Enclosure

Temperature of the lubricant in the differential was controlled through an electric air heater and forced convection to increase temperature, or water spray and evaporative cooling off of the housing to lower it as required.

Two types of operation were conducted on the HMMWV differential. The first was a defined cycle based upon vehicle data supplied by TARDEC to ARL. This cycle, as modified to run on the available laboratory equipment, is shown in Figure 4 (next page).

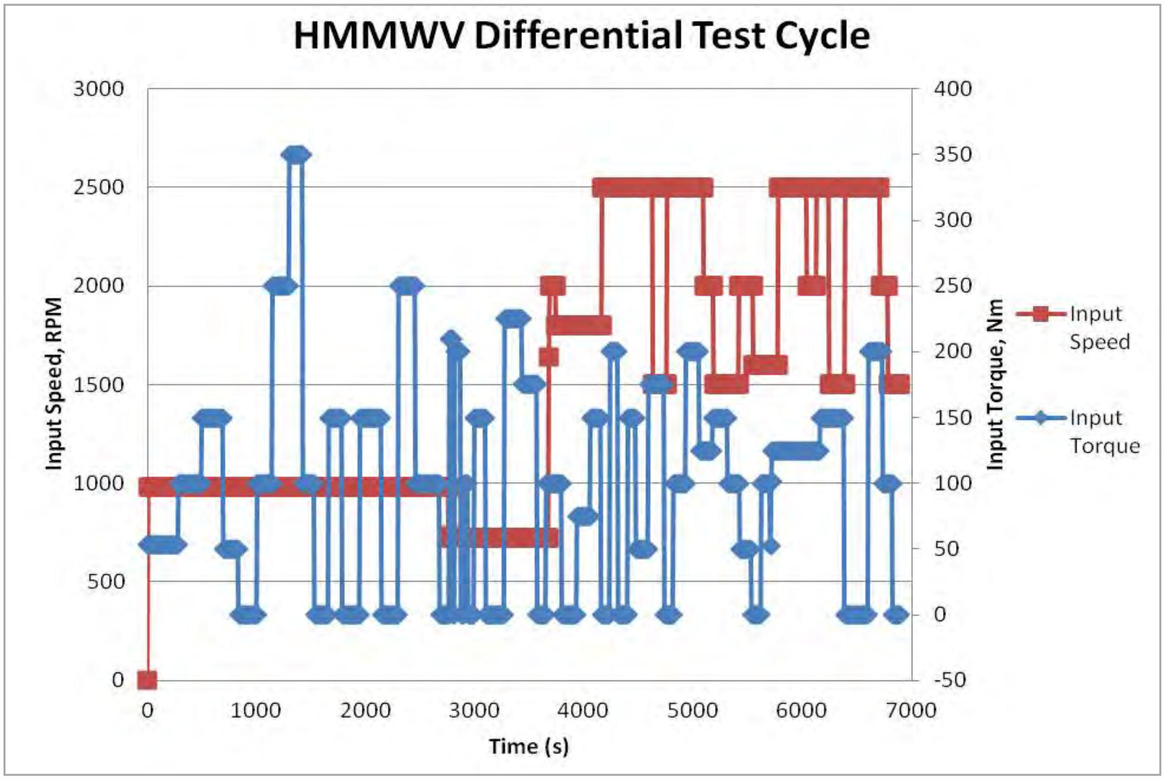

Figure 4: HMMWV Axle Drive Cycle

The second cycle consisted of a set of steady state conditions conducted at two controlled oil temperatures of 80 and 100 °C. Operation was conducted at four input speeds (700, 1100, 1700, and 2500 RPM) over five input torques (50, 100, 200, 300, 400 Nm).

2.2 HMMWV Stand Test Fluids

Three oils were used for the stationary stand evaluations. The first two were the baseline and candidate axle oils from the initial FMTV SAE J1321 testing conducted [1], and included the same SAE 80W-90 and a synthetic SAE 75W-140 oil. The third lubricant was selected because of its viscosity properties and was a universal tractor lubricant meeting the SAE 80 high temperature viscosity properties. It is referred to in the following sections as “xxW80”.

2.3 EFFICIENCY TESTING RESULTS

Efficiency measurements were based upon input torque and the sum of the output shaft torques. Figure 5 shows the differential efficiency and fluid temperature for each of the three lubricants over the drive cycle.

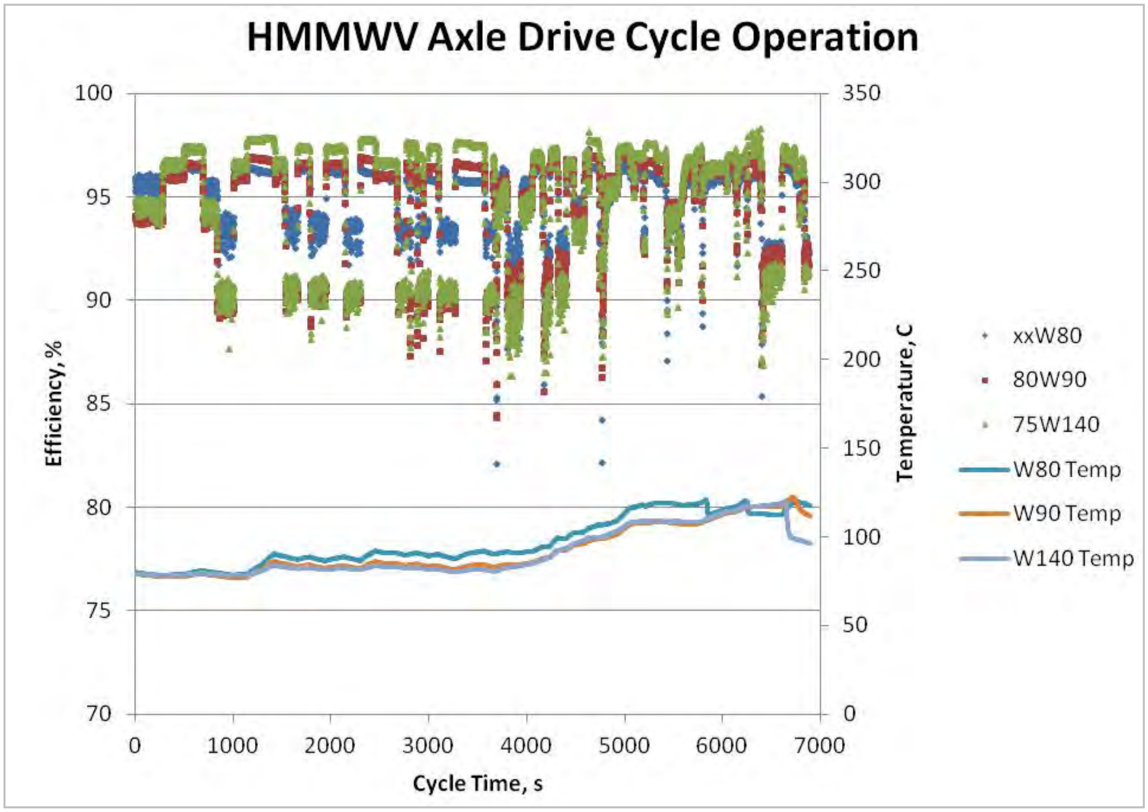

Figure 5: HMMWV Drive Cycle Operation

For the driving cycle, 57 points were identified for a weighted efficiency analysis. While there are more distinct points than this over the course of the cycle, the transient nature of some of the operating points did not allow for stabilization of the test component, and thus were excluded. To compare one lubricants performance to another, the power lost through the differential at each of the 57 points was then evaluated. This caused a greater emphasis to be placed on the efficiency at higher power transmission conditions. A difference of efficiency from 95 to 98% when input power is 5hp does not have as much absolute benefit as the same percentage improvement at a 50hp condition. The loss at each step was summed to produce an overall cycle power loss. The

value for each of the three oils, and relative change with respect to the 80W-90 baseline oil is shown in Table 1.

Table 1: Cycle Power Loss Values

	80W-90	xxW80	75W-140
Cycle Power Loss (kW)	36.85	39.91	33.83
Relative Change		7.67%	-8.92%

These results indicated that there is a potential for reduction in differential based power loss through the use of the SAE 75W-140 gear oil. While it can be seen in Figure 5 that no particular oil has the best efficiency over the entire operating range, the SAE 75W-140 tended to outperform the other two during the more highly loaded portions of the cycle.

Efficiency values for the steady state load points are shown in Table 2 through Table 7.

Table 2: 80W-90 Steady State Efficiency at 80 °C

		Input Torque (Nm)				
		50	100	200	300	400
Input RPM	700	95.52	96.43	96.63	96.60	96.45
	1100	95.07	96.42	96.98	97.04	96.92
	1700	93.51	95.85	96.98	97.25	97.30
	2500	92.36	95.52	97.20	97.41	97.49

Table 3: xxW80 Steady State Efficiency at 80 °C

		Input Torque (Nm)				
		50	100	200	300	400
Input RPM	700	94.75	95.53	95.68	95.63	95.68
	1100	95.04	96.06	96.29	96.30	96.25
	1700	94.16	95.98	96.71	96.79	96.77
	2500	92.56	96.04	96.90	97.13	97.17

Table 4: 75W-140 Steady State Efficiency at 80 °C

		Input Torque (Nm)				
		50	100	200	300	400
Input RPM	700	96.57	97.34	97.72	97.67	97.56
	1100	95.47	97.06	97.80	97.89	97.87
	1700	93.56	96.18	97.55	97.92	97.98
	2500	92.59	95.77	97.31	97.81	97.99

Table 5: 80W-90 Steady State Efficiency at 110 °C

		Input Torque (Nm)				
		50	100	200	300	400
Input RPM	700	95.35	95.97	96.00	95.91	95.75
	1100	95.47	96.32	96.52	96.44	96.43
	1700	95.16	96.47	96.92	96.99	96.86
	2500	93.90	96.11	97.01	97.17	97.15

Table 6: xxW80 Steady State Efficiency at 110 °C

		Input Torque (Nm)				
		50	100	200	300	400
Input RPM	700	94.44	95.11	95.37	95.40	95.23
	1100	94.77	95.56	95.89	95.86	95.78
	1700	94.65	95.82	96.29	96.23	96.13
	2500	93.70	95.61	96.40	96.41	96.61

Table 7: 75W-140 Steady State Efficiency at 110 °C

		Input Torque (Nm)				
		50	100	200	300	400
Input RPM	700	96.30	96.96	97.15	97.01	96.84
	1100	96.03	97.11	97.56	97.58	97.46
	1700	95.44	96.97	97.73	97.86	97.86
	2500	93.97	96.28	97.50	97.86	97.95

The data from Tables 2 through 7 are presented as graphs in Figures 6 and 7.

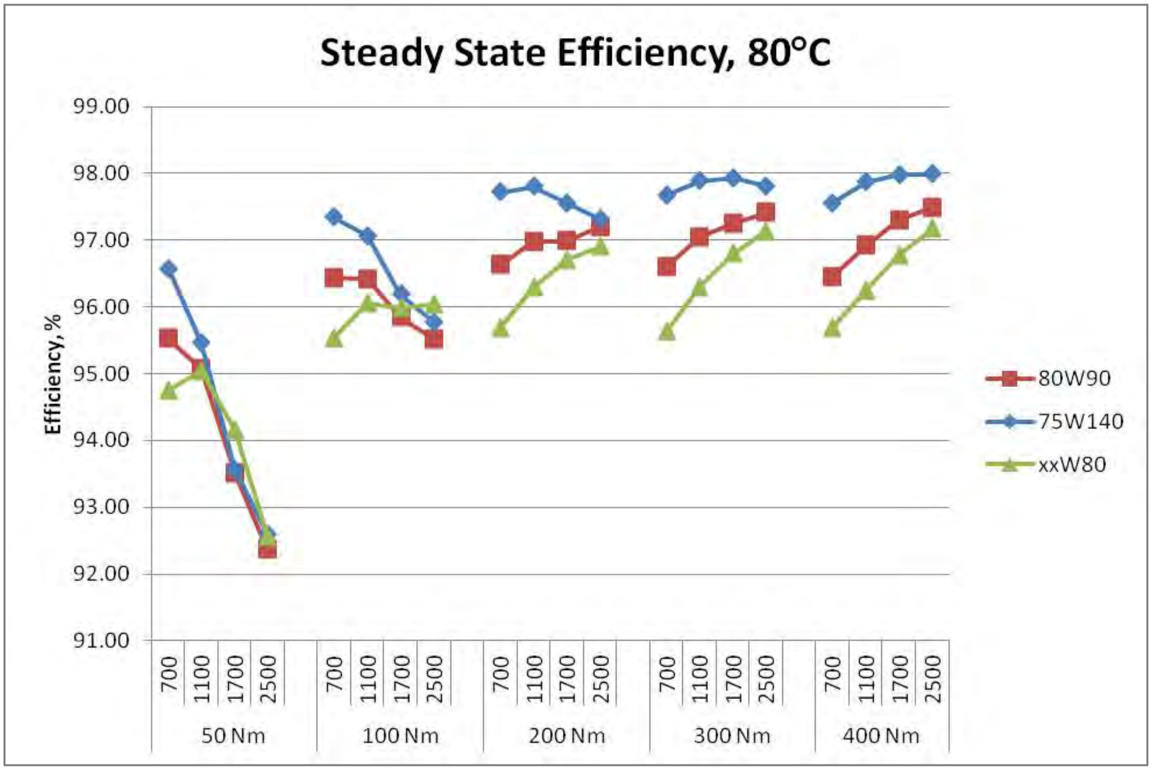

Figure 6: Steady State Efficiency at Controlled Temperature of 80 °C

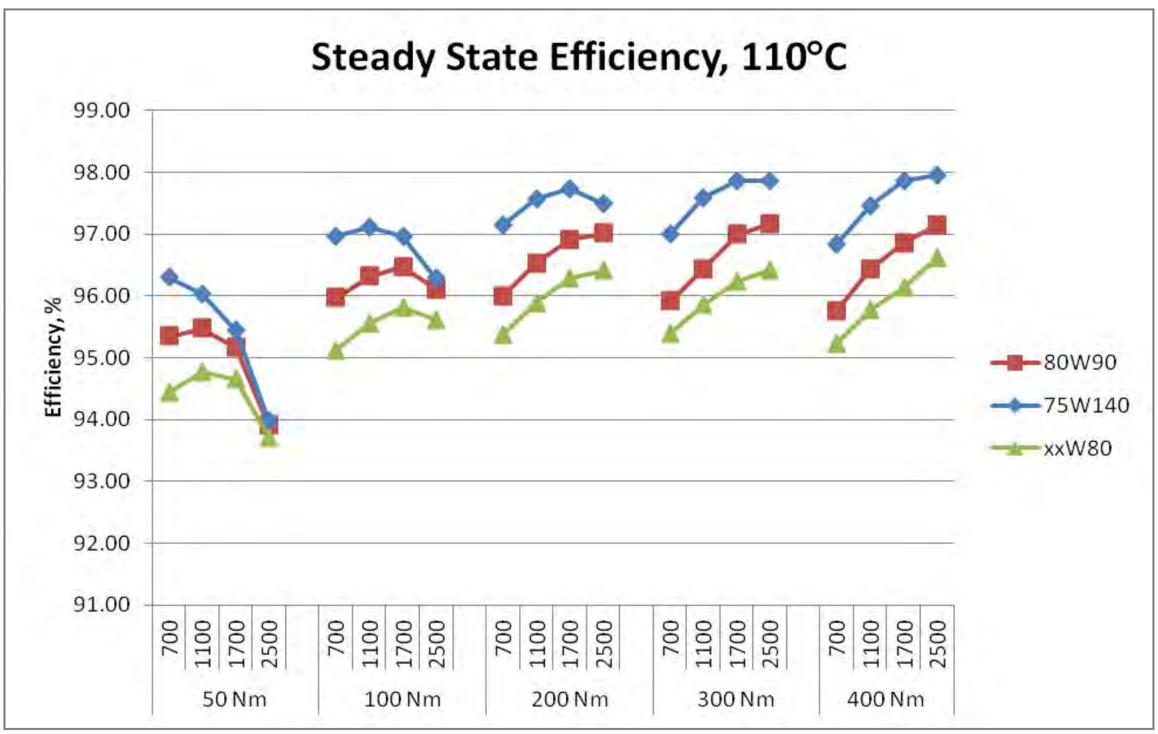

Figure 7: Steady State Efficiency at Controlled Temperature of 110 °C

The data indicates that the efficiency of a given oil is a function of the load, speed, and operating temperature. As the speed increased at low torque levels, all three lubricants experienced a reduction in efficiency due to churning losses making up a higher percentage of power absorption. However, as speed was held constant and load increased, resulting in higher gear tooth contact forces, the ability of the higher viscosity oils to maintain film thickness and reduce friction becomes apparent. These experimental observations connecting fluid temperature and efficiency are in agreement with those previously seen in modeling simulations [11].

3.0 FUTURE STATIONARY AXLE TEST METHOD DEVELOPMENT

Based upon the results from HMMWV differential testing, it was determined that discriminating between two oils for efficiency was possible using a motored, stationary test stand. A drawback to this method was that results are not directly tied to a vehicle fuel consumption change, but rather a component efficiency. By utilizing the field data produced through SAE J1321 testing and simulating the cycle in full-vehicle laboratory conditions, a connection between the two can begin to be formed.

3.1 COMPONENT SELECTION

The wide range of equipment utilized by the military complicates the process of selecting a gear oil for efficiency benefits. To determine if the effects in one size range trend in the same direction for others, a light-, medium, and heavy-duty axle were identified for future evaluation. Table 8 lists a selection of common Army equipment and their associated axle information.

Table 8: Common Army Equipment Axle Information

Vehicle	Axle Make	Drive Axles	Axle Ratio	Differential
M1097 HMMWV (Stationary Test Item)	-	2	5.24:1	Hypoid
M1083A1P2 FMTV (SAE J1321 Test Vehicle)	Meritor RF-19-611 (Front) Meritor RT-15-611 (Rear)	2 or 3	7.8:1	Amboid
M1074/M1075 PLS	AxleTech (formerly Rockwell SVI 5MR)	5	6.0:1	Spiral Bevel
M1070 HET	AxleTech (formerly Rockwell)	4	7.36:1	Spiral Bevel
M1070A1 HET	AxleTech 5000 Series	4	6.945:1	Spiral Bevel
RG33	AxleTech 5000 Series	3	7.56:1	Spiral Bevel
RG31A2	AxleTech 4000 Series	2	-	Spiral Bevel
BAE Caiman	AxleTech 4000 Series	3	6.14:1	Spiral Bevel
MaxxPro	AxleTech 5000 Series	2	Front: 6.14:1 Rear: 5.86:1	Spiral Bevel

In selecting an axle of each size range, the HMMWV and M1083A1P2 FMTV options had the appeal of having been used for initial investigations. This made them ideal candidates for continued use as a light- and medium-duty option. For the heavy-duty size range, there are many similarities between the current M1070 HET and M1074/M1075 PLS [9,10]. Both utilize a spiral bevel differential and planetary wheel-end hub reduction, and encompass a wider selection of overall axle ratios, the Palletized Load System (PLS) offered a final drive ratio between that of the HMMWV and FMTV, and vehicle data was anticipated from the TARDEC (Power and Energy Vehicle Environmental Laboratory) PEVEL. Thus it was selected for testing. While the MRAP vehicles did present additional options during selection, their axle sizing fell more into the light and medium size range than the desired heavy duty range. As a result, the following axles were selected for future stationary testing to represent a wide spectrum of the military fleet , and best utilizing the data already produced:

- Light-duty: HMMWV Rear Axle Differential and Wheel-end Hubs
- Medium-duty: M1083A1P2 FMTV Rear Tandem, Rear Axle
- Heavy-duty: M1074 PLS Rear Tridem, Rear Axle

3.2 FMTV CYCLE RECREATION

During the SAE J1321 testing at Pecos, TX [2], an extensive log of parameters were monitored and logged through the vehicles CAN bus system. To be able to recreate the operating cycle for the stationary stand, the logged data was reviewed, and the parameters of interest identified were the engine speed, engine load at speed, transmission gear, transmission output shaft speed, vehicle speed, and the torque converter lockup status. Using this data, the engine speed and load for each given point in the cycle was then determined by combining vehicle data acquired during SAE J1321 testing and engine power output maps for the Caterpillar C7 from previous engine dynamometer tests.

For calculating axle input speed, the vehicle speed, tire diameter, and gear ratios were used to back calculate the input requirement. The overall driving cycle from the SAE J1321 (based on distance) can be seen in Figure 8. Using the speed, and the above mentioned constants, the input speed was identified, and is shown in Figure 9.

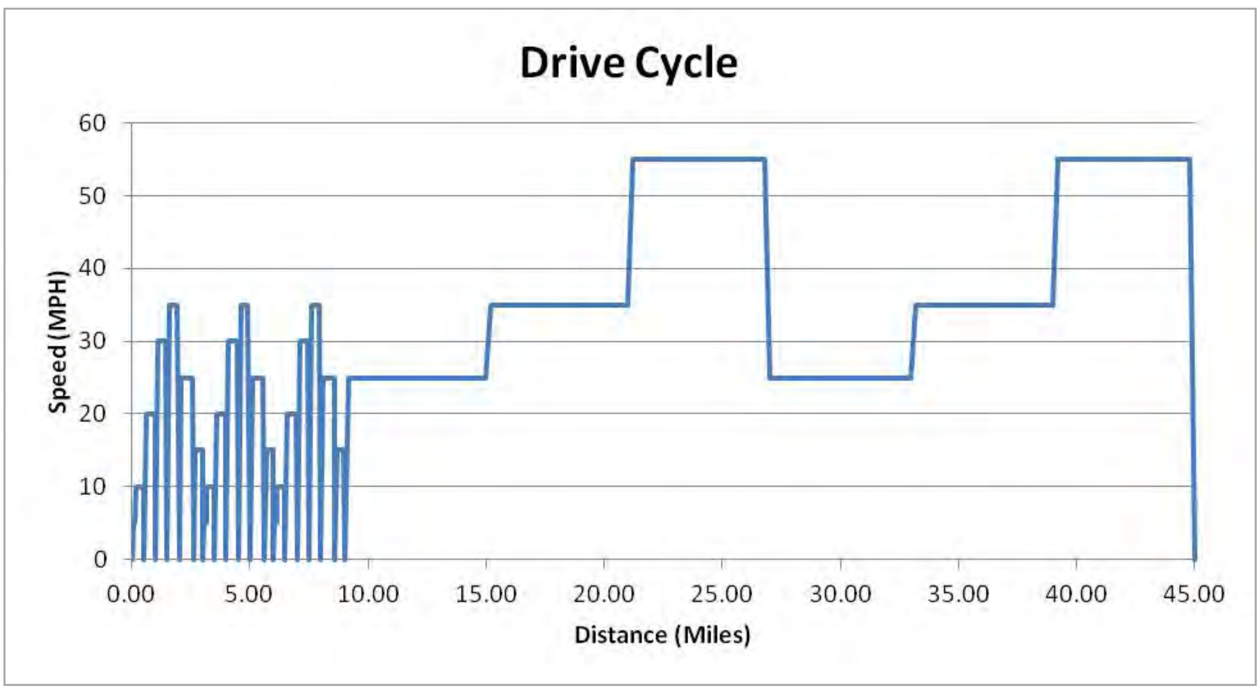

Figure 8: Drive Cycle

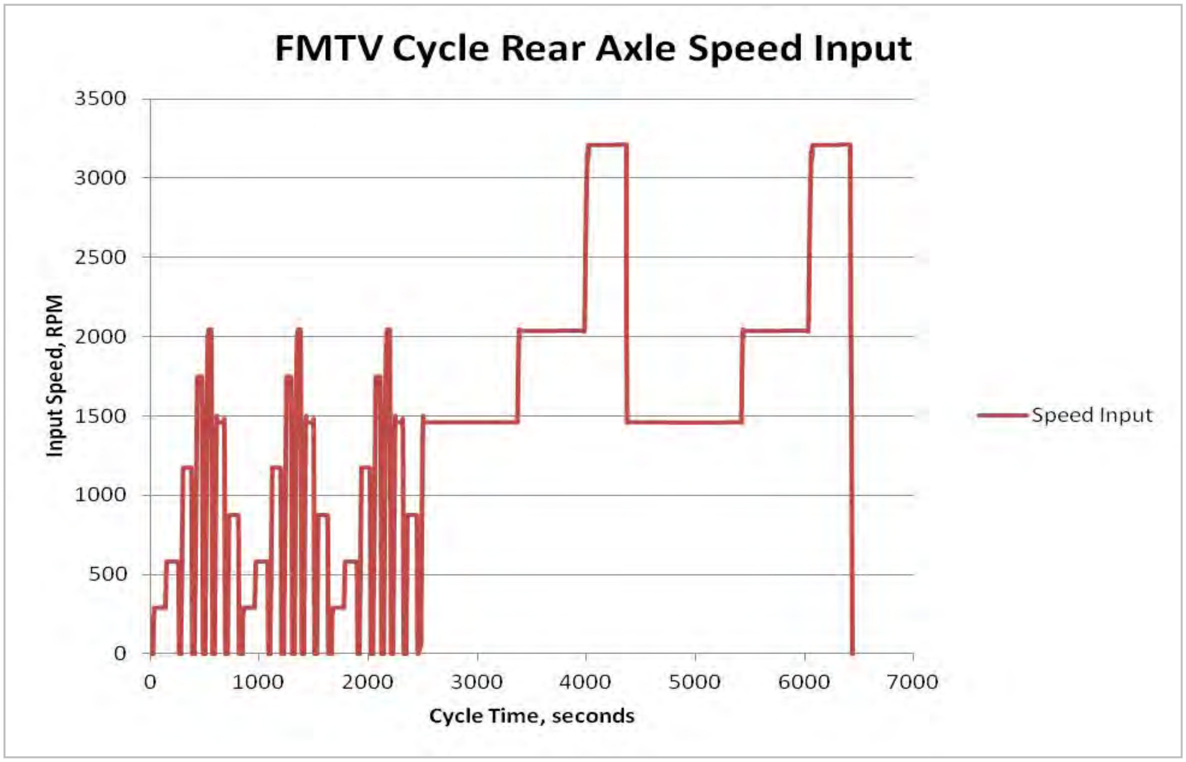

Figure 9: FMTV Cycle Rear Axle Speed Input

The axle torque input calculations are more complicated than that of speed. First it was decided that axle #3 was the most appropriate for stationary laboratory testing. By selecting the rear axle from the tandem, the inter-axle differential is eliminated, and the articulated ends of the front steering axle are eliminated, thus simplifying hardware installation. It is known that the FMTV transfer case proportions the output power 30% to the front axle and 70% to the rear tandem, so resulting input torque to each of the rear axles is estimated at 35% of the total engine torque output (ignoring losses). Without instrumentation on the intermediary shaft connecting the rear tandem axles, which was outside the scope of the previous SAE J1321 test program, estimated values were determined based upon known C7 power characteristics, and the acquired CAN bus data. Using the "engine percent load at current speed" parameter (J1939 SPN 92), along with laboratory data of full load power curves for the C7 engine, the power output of the engine was estimated over the drive cycle. From there the current gear ratio of the transmission was used to calculate the output torque and speed leaving the transmission. The torque was then scaled to

estimate the per axle input torque. The equations used to calculate the per axle input torque are shown in Figure 10.

$$HP(S) = PR(S) * HP_{max}(S)$$

Where

S	=	Current engine speed (rpm)
PR	=	Engine percent load at speed, S
HP_{max}	=	Maximum engine horsepower at speed, S Derived value using third order polynomial generated from TFLRF Caterpillar C7 power curves
HP	=	Estimated engine horsepower at speed, S

$$T_{Engine}(S) = (HP(S) * 5252)/S$$

Where

T_{Engine}	=	Estimated engine torque at speed, S

$$T_{TransOut}(S) = T_{Engine}(S) * TGR$$

$$S_{TransOut}(S) = S/TG\,R$$

Where

TGR	=	Transmission gear ratio at speed, S
$T_{TransOut}$	=	Estimated transmission output torque at speed, S
$S_{TransOut}$	=	Transmission output speed and engine speed, S

$$T_{AxleIn}(S) = T_{TransOut}(S) * 35\%$$

Where

T_{AxleIn}	=	Single rear axle input torque at speed, S

Figure 10: FMTV Axle Input Torque Calculations

The transmission output speed was compared back to the previously calculated axle input speed for verification. Final resulting torque input to the rear axle of the vehicle is shown in Figure 11.

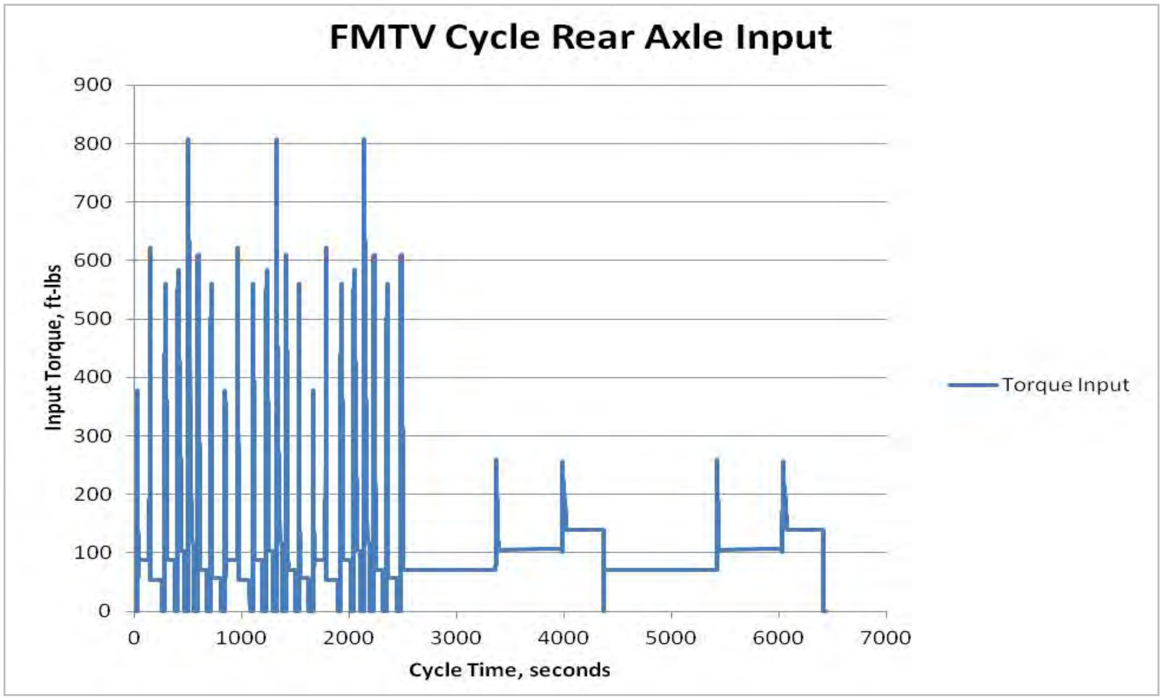

Figure 11: FMTV Cycle Rear Axle Torque Input

It should also be noted that no torque multiplication factor was included in the axle input torque calculations from torque convertor multiplication for the stationary stand. Torque multiplication, although potentially high, occurs only at very high differential stator and turbine speeds in the torque convertor and reduces drastically as the vehicle attains speed. For recreating the drive cycle of the stationary stand, the focus of this data set was more on the longer duration steady state conditions.

Since the vehicles were driven on dry pavement, there was also an assumption made that no wheel-slip was occurring, resulting in typical power distribution through the transfer case and intermediate axle. Some minor differential action would have been occurring due to the circular nature of the track, but due to 9-mile circumference the outer wheel would only be traveling 0.106% faster than the inner tire. Because of this, differential action was not considered for the stationary stand test development.

Based upon the input speeds and loads shown above, a stationary axle test is expected to differentiate between gear oils in a manner similar to that seen in the full vehicle. While the input requirements are shown in the figures above, another appropriate method of control for the cycle would be to determine desired output parameters for speed and torque at the wheel hubs and adjust input power to meet these. This was the approach taken for the heavy duty axle since no real world vehicle data exists.

3.3 PLS DRIVE CYCLE

3.3.1 PLS Simulation Data Processing Overview

The PLS simulation was done by TARDEC using the same SAE J1321 drive cycle provided by SwRI as used for the FMTV's. This drive cycle was designed to simulate a combination of stop-and-go driving along with limited duration medium and high speed operation. It was based upon two cycles from SAE J1376: the "Local Test Cycle" and the "Short Haul Test Cycle." A graphical representation of the drive cycle is shown in Figure 8. The drive cycle was used by TARDEC to input into a PLS vehicle simulation, so that output wheel torque could be identified. Once completed the data was processed by SwRI to verify it simulated the drive cycle correctly, and also to determine the required axle input torque and speed from the simulated axle output torque and speed. The method for achieving this is described below.

3.3.2 Drive Cycle Verification

The vehicle velocity from the PLS simulation data was converted from km/hr to mi/hr and was plotted versus distance. This was then compared to the actual drive cycle vehicle speed versus distance in order to verify the drive cycle speeds and position at a given speed were properly simulated. This is shown below in the Figure 12. As seen, the PLS drive cycle simulation was verified to the original drive cycle supplied.

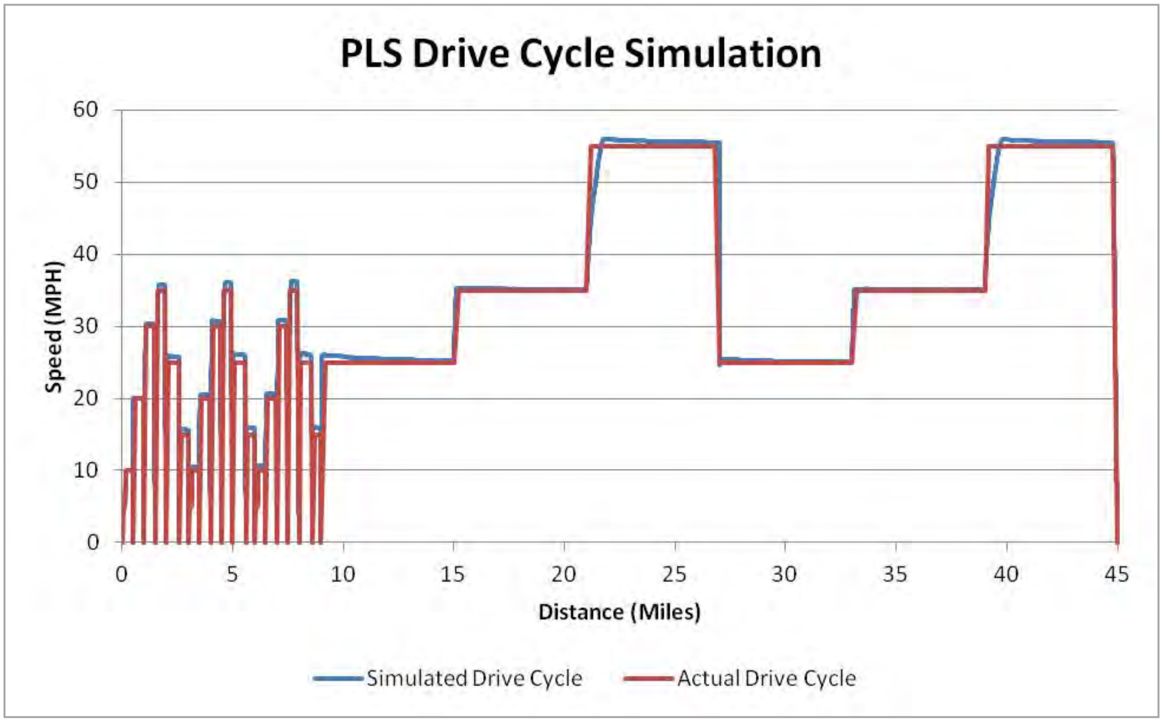

Figure 12: Comparison of Simulated and Actual Drive Cycles

3.3.3 PLS Axle Torque – Speed Requirements

There were 34 data points from the PLS simulation which were of significantly higher torque values than all other data points. These high torque points occurred during the initial start and during some gear changes, and would require special costly test stand equipment to replicate. Since the test stand will evaluate steady state conditions (which were found at much lower nominal torque values) these points (34 out of 6293 points) were removed from use on the stationary cycle. Figure 13 shows the unmodified torque points and Figure 14 shows the modified torque points versus vehicle speed from the PLS simulation.

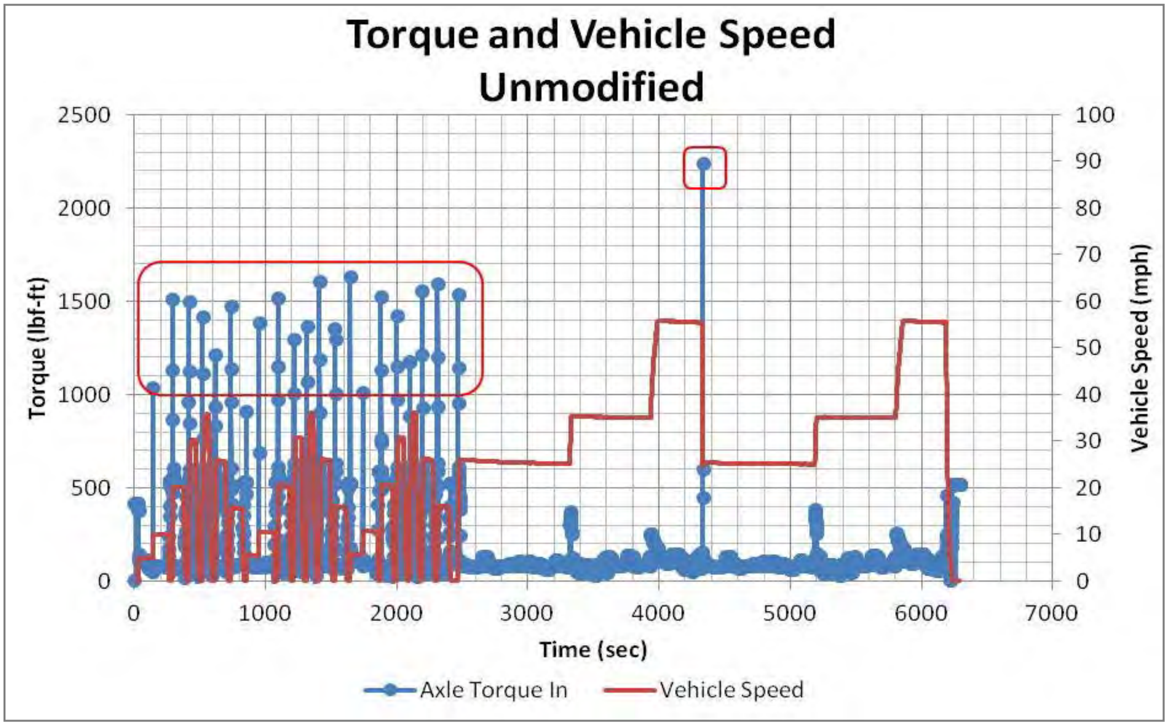

Figure 13: PLS Torque and Vehicle Speed, All Points

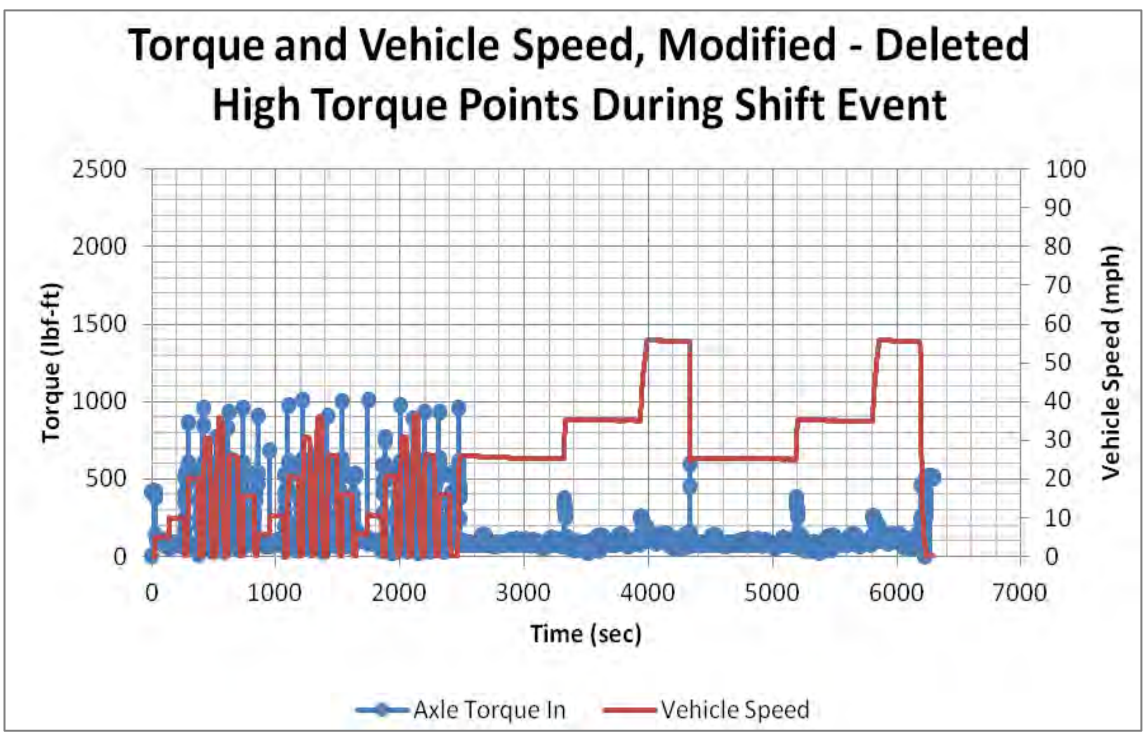

Figure 14: PLS Torque and Speed, Modified

The three left and three right rear wheel torques from the simulation data were then averaged for every time step (1 second) to estimate the average wheel torque for each of the three rear axles. This number was then multiplied by 2 in order to find the total axle output torque, and for use in back calculating the input torque required at the axle input. These torque output data sets were plotted versus output speed to generate a torque-speed curve for the axle output. This curve was used to size the stationary test stand gearboxes and dynamometer. The axle total output torque-speed curve is shown in Figure 15 below.

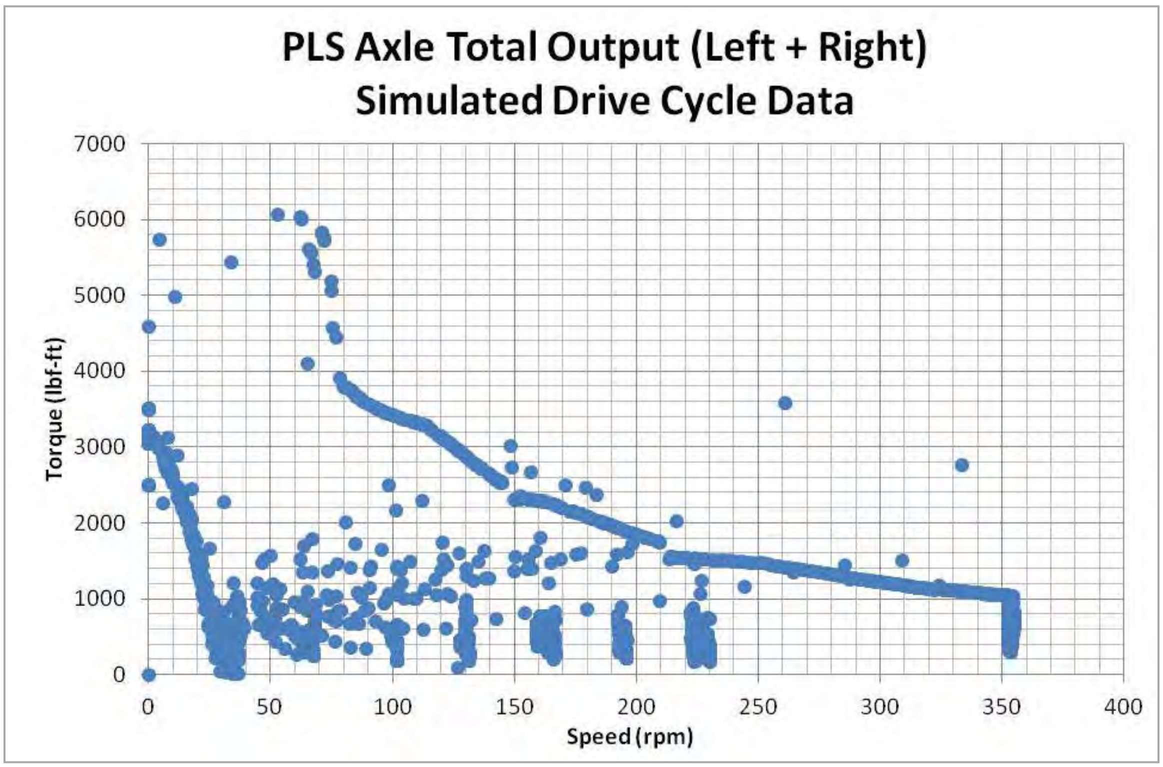

Figure 15: PLS Total Axle Output Torque and Speed

The total output torque was divided by the overall axle ratio, 6, to calculate the input torque. The wheel speed was multiplied by the overall axle ratio, 6, to calculate the input speed. A similar torque-speed curve for the input side was then generated using this data and used to size the required input motor for the stationary stand. The axle input torque-speed curve is shown in Figure 16.

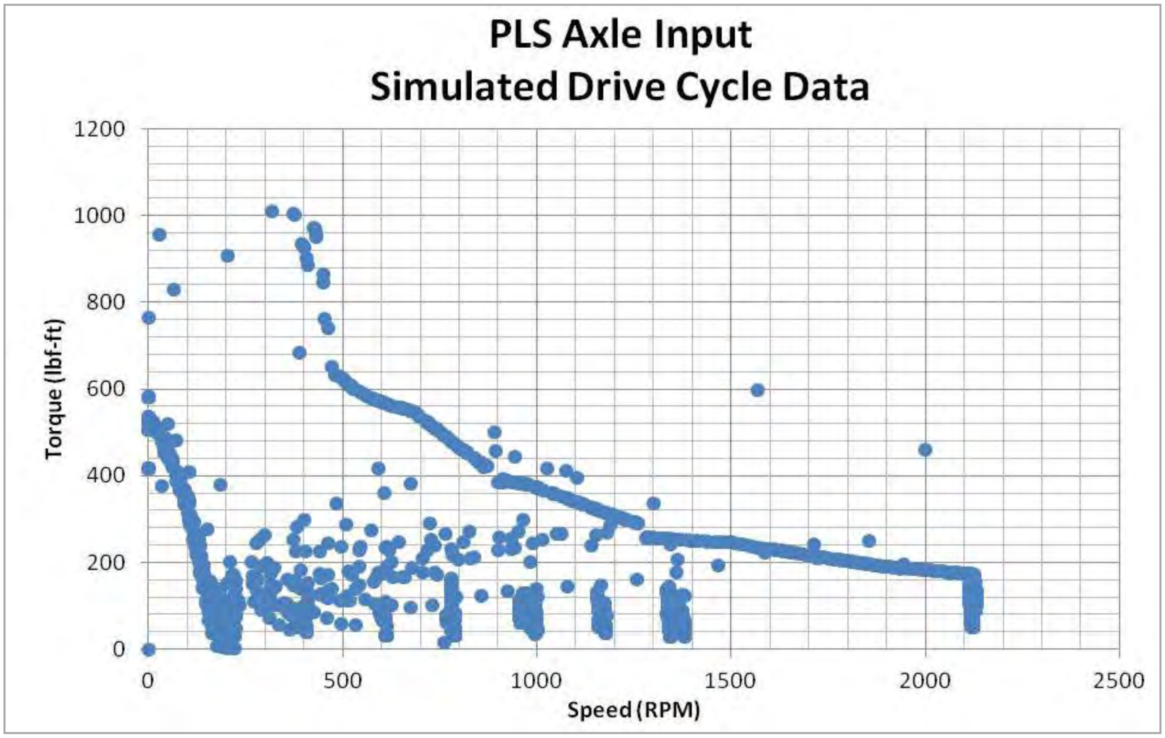

Figure 16: PLS Axle Input Torque and Speed

3.4 APPLICATION TO LIGHT-DUTY EQUIPMENT

Since the drive cycle used for the FMTV testing and PLS simulation is derived from a vehicle speed-distance profile, adapting it to other axles is possible with knowledge of major vehicle components. The calculation of wheel, and therefore axle input, speed is determined based upon the vehicle velocity. If it is assumed that no slippage is occurring between the wheel and road surface, the required axle speed input is a function of the tire diameter, geared hub ratio (if present), and differential ratio. This is shown in the equation below.

$$\omega_I = \frac{v \times 16.667}{2\pi \times r} \times R_H \times R_D$$

$$v = Vehicle\ Speed, kph$$

$$r = Tire\ Radius, meters$$

$$R_H = Ratio\ of\ the\ Geared\ Hub$$

$$R_D = Ratio\ of\ the\ Differential$$

$$\omega_I = Differential\ Input\ Speed, RPM$$

The determination of axle load becomes more complicated without operational field data to draw from for the desired cycle. The preferred method of data acquisition would be direct instrumentation of drive shafts on a vehicle; however this may not be feasible in all cases. A secondary method, when available, would be to use road load factors for a vehicle to estimate the required wheel torque to accelerate and maintain a given speed. While these are typically available for light-duty passenger vehicles due to EPA emissions testing requirements, they are not as readily available for the heavy-duty market or military vehicles. Due to this, vehicle data should be used if at all possible for determination of an axle loading profile for a specific vehicle like the HMMWV. If available, as in the case of the SAE J1321 testing with FMTVs, CAN based vehicle data could also provide a usable estimation of the vehicle power and torque levels.

3.5 TEST FACILITY AND EQUIPMENT CONSIDERATIONS

In setting up a facility to conduct stationary axle testing for military purposes, there are a few considerations which must be taken into account. Unlike the standardized axle tests listed in SAE J2360, the components of interest for military equipment often contain wheeled-hub reductions utilizing the same gear lubricant. This results in an entire axle assembly being required for testing rather than just a differential. For the heavy-duty military equipment, M1070 HET and M1074 PLS size range, an axle assembly can reach up to 96 inches across between the wheel mounting flanges. This is considerably wider than the Dana Model 60 used in the L-37 test, resulting in expanded layout requirements for conducting evaluations [12].

Motoring and power absorption equipment should be sized based on the largest axle to be tested. While it is common in standardized tests to use an engine driven stand and transmission to supply power to test components, an electric motor with appropriately sized variable frequency drive can provide the low speed torque produced by lower transmission gears. This can be seen in Figure 17 which shows the speed and load inputs for the FMTV sized cycle versus the torque curve of a 250HP, 1200 RPM base speed electric motor.

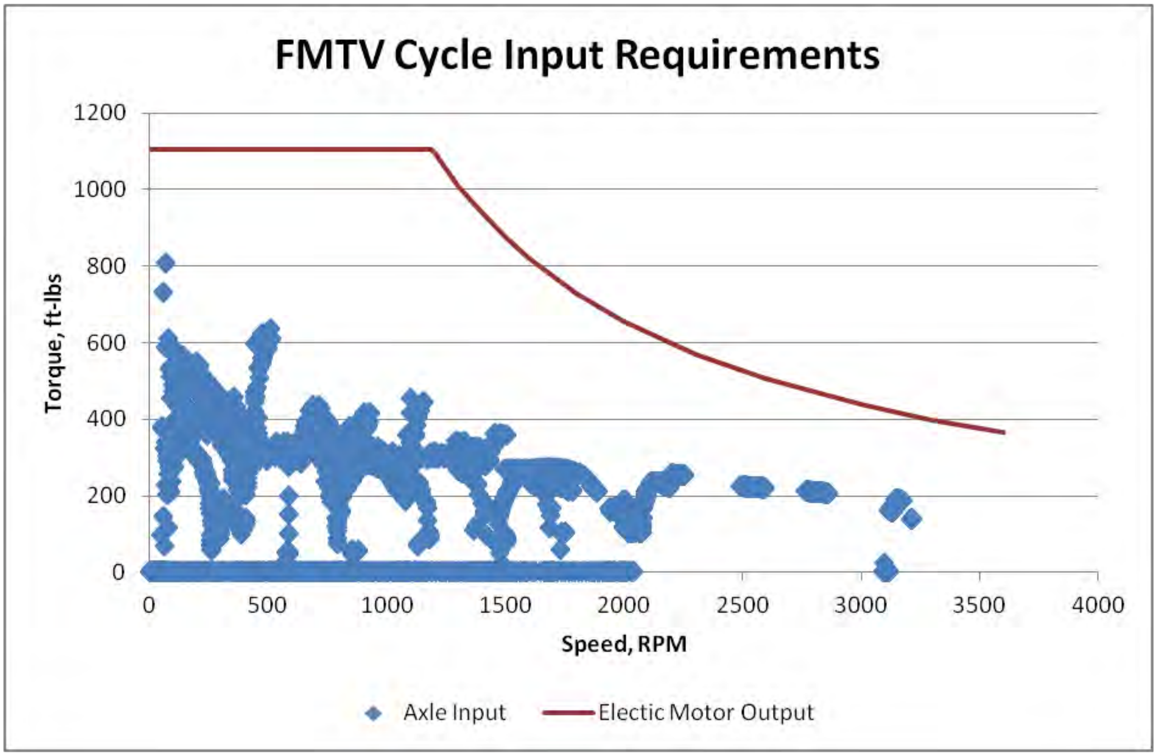

Figure 17: FMTV Cycle Input Requirements

On the output side of the axle, the low speed torque over power output is the driving factor in dynamometer sizing more. Since it would be desirable to test multiple sizes and ratios of axles within the same facility, an appropriate method of absorption would be to connect the wheel hub output to a torque reducer. This would bring the input speed to the absorbing dynamometer into a range which the load can be better controlled and absorbed. An example layout of major components is shown in Figure 18.

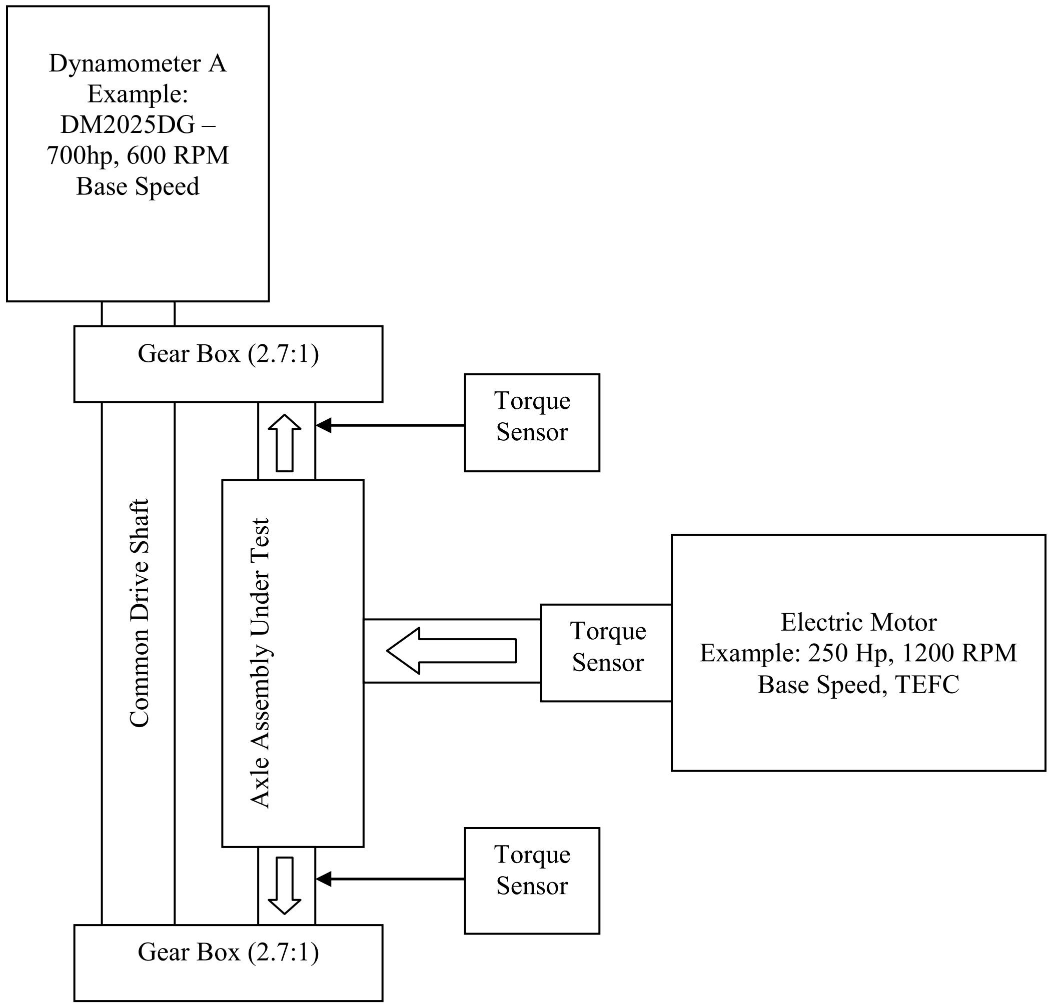

Figure 18: Axle Test Stand Layout

4.0 CONCLUSION AND RECOMMENDATIONS

Based upon the measured changes in fuel consumption for in-vehicle testing, there are significant potential savings associated with advanced powertrain lubricants [1,2]. A vehicle level improvement in the 6-7% range with no required hardware changes provides an appealing reduction in fuel, logistical, and financial burdens for the U.S. Army. Future investigation into lower viscosity gear oils may produce additional fuel consumption benefits, but must be balanced with ensuring adequate protection is provided for internal contact surfaces. Laboratory tests should be utilized for this purpose.

Since the axle is typically cooled only through forced convection during operation, the energy balance reached through efficiency and heat loss determines how a fluid impacts fuel consumption. If a fluid is too low in viscosity, inadequate film thickness may result in increased friction and heat while at the same time result in decreased churning losses in the bulk fluid. A higher viscosity fluid may heat from the bulk churning, but keep localized gear temperatures lower due to an improved film thickness. In a laboratory setting, the ability to control external cooling and internal loading is much greater than full-vehicle testing conducted in the field. This should allow for a range of operating conditions and temperature to be isolated in identifying if a duty-cycle specific lubricant must be used, or if a common fluid will work for all vehicle types.

It has been shown that it is possible to differentiate between lubricants based on efficiency over both a drive cycle and steady state conditions in a light duty application using a HMMWV differential, providing a background for the future use of military equipment in this type of evaluation. Combined with the existing bench top and other lab style evaluations for ensuring proper gear protection in commercial hardware, the use of laboratory scale efficiency screening to select improved lubricants may be beneficial to the U.S. Army.

Future work that should be considered includes:

- Exploration of light, medium, and heavy-duty correlation for efficiency improvements
- Gather additional vehicle data of the described cycle for stationary test development

- Determination of laboratory method repeatability
- Conduct additional steady state operation allowing differential to reach a stabilization temperature based upon ambient conditions

With the use of synthetic base stocks and advanced additive technologies, there is potential for fuel saving effects and extended drain intervals throughout ground vehicle powertrains.

5.0 REFERENCES

1. Warden, R.W., Frame, E.A., Brandt, A. C., "SAE J1321 Testing Using M1083A1 FMTVS", Interim Report TFLRF No. 404, March 2010.
2. Warden, R.W., Frame, E.A., Interim Report TFLRF No. 444, "Axle Lubricant Efficiency", May 2014
3. Fuel Consumption Test Procedure - Type II, SAE J1321, 2012
4. Technical Manual Operator's Instructions: M1083 Series, 5 TON, 6x6, Medium Tactical Vehicles (MTV), TM-9-2320-366-10-1, 1998
5. Lubricating Oil, Gear Multipurpose (Metric) Military Use, J2360, 2008
6. Technical Manual Lubricating Oil, Internal Combustion Engine, Combat/Tactical Service, MIL-PRF-2104H, 2004
7. Brandt, A.C., et. al., "Feasibility of Using Full Synthetic Low Viscosity Engine Oil at High Ambient Temperatures in Military Vehicles," SAE Technical Paper 2010-01-2176, 2010
8. Fuel Economy Measurement Test (Engineering Type) for Trucks and Buses, SAE J1376, 1982
9. Technical Manual Operator's Manual for Truck, Tractor, 8x8 M1070A1, TM-9-2320-427-10, January 2011
10. Technical Manual Operator's Manual for Truck TRUCK, TRACTOR, M1074 AND M1075 PALLETIZED LOAD SYSTEM (PLS), TM 9-2320-364-10, April 2009
11. Kolekar, Anant S., Andrew V. Olver, Adam E. Sworski, and Frances E. Lockwood. 203-209. <www.elsevier.com/locate/triboint>.
12. ASTM Standard D6121, 2011, "Standard Test Method for Evaluation of Load-Carrying Capacity of Lubricants Under Conditions of Low Speed and High Torque Used for Final Hypoid Drive Axles," ASTM International, West Conshohocken, PA, 2011, DOI: 10.1520/D6121-11